40th Turbomachniery Symposium
Abstract

Coupled Sub-Synchronous Vibration of Lateral and Axial Directions

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Classification of Sub-Synchronous Vibration

Sub-Synchronous Vibration

Self-excited Vibration (Unstable Vibration) $\zeta < 0$

-Oil whip
(Unstable force of sleeve bearing)

-Steam whirl
(Labyrinth seal, torque unbalance)

-Gas whirl
(Labyrinth seal, impeller stable force)

-Hydro whirl
(Annular seal, impeller stable force)

-Internal friction
(Damping on rotor)

-Asynmetric shaft
(Asynmetric of stiffness or inertia)

Forced Vibration

Small Damping Ratio Case $\zeta \geqq 0$

-Hydraulic disturbance response
(Low flow operation)

-Mechanic disturbance response

-Electric disturbance response

Enough Stability Case $\zeta > 0$

-Surging, Reverse flow

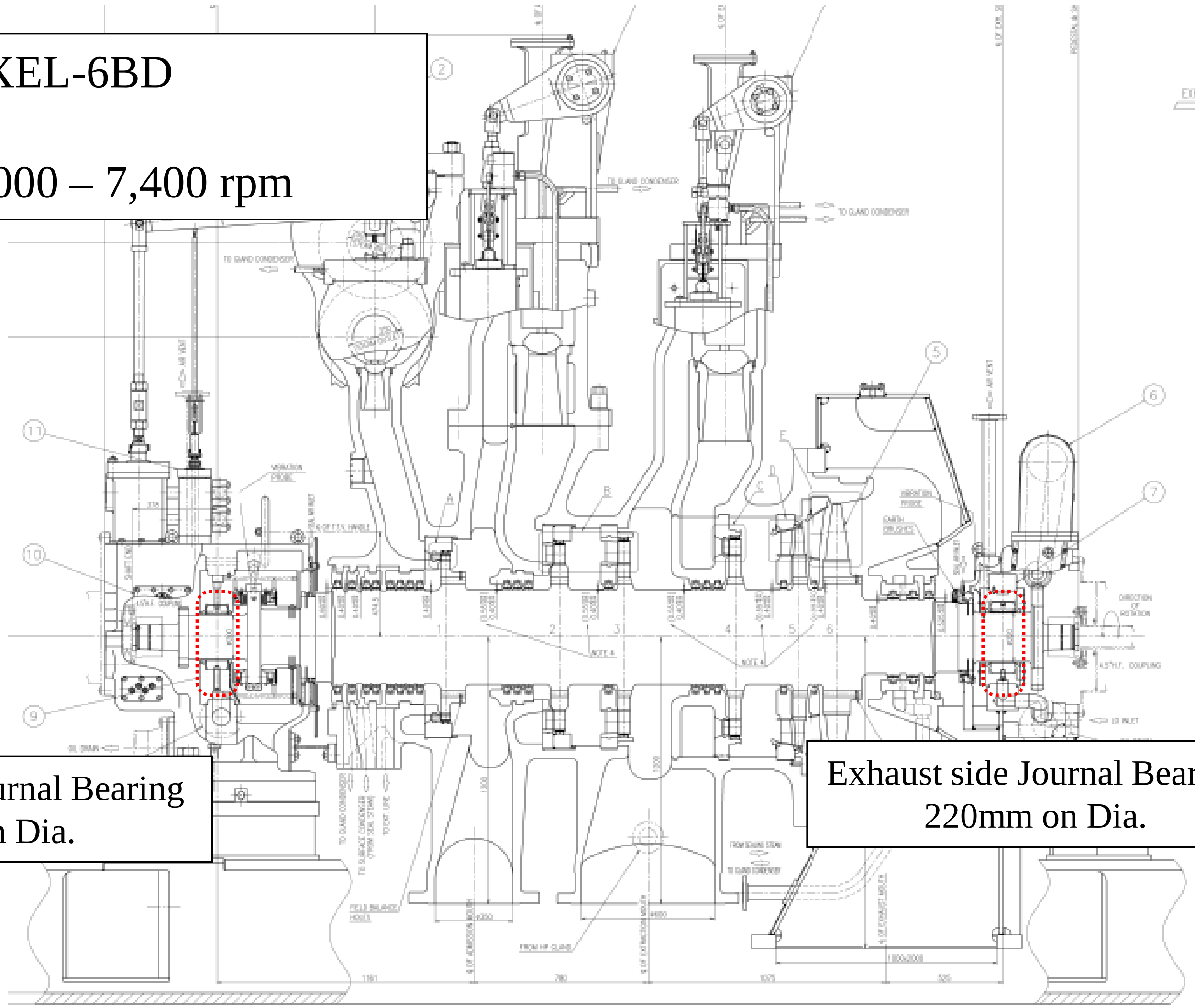
-Karman Vortex
(Resonance with Karman Vortex)

Coupled Sub-Synchronous Vibration Lateral and Axial Directions

Case-1

During the shop mechanical running test of the following steam turbine, the sub-synchronous vibration was observed.

Turbine Model: 7MXEL-6BD
Power: 26MW
Operating Range: 5,000 – 7,400 rpm



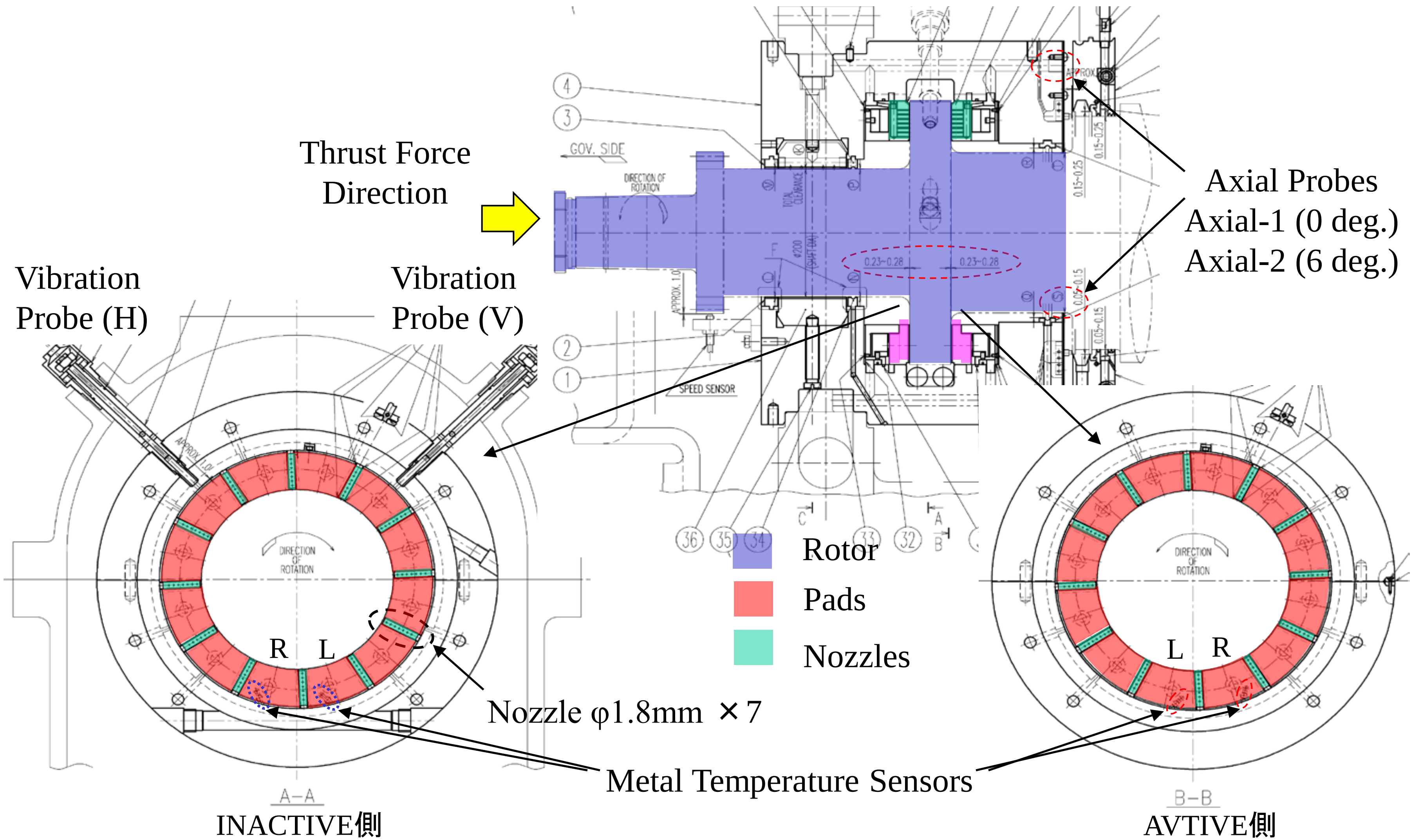
Governor side Journal Bearing
200mm on Dia.

Exhaust side Journal Bearing
220mm on Dia.

Coupled Sub-Synchronous Vibration Lateral and Axial Directions

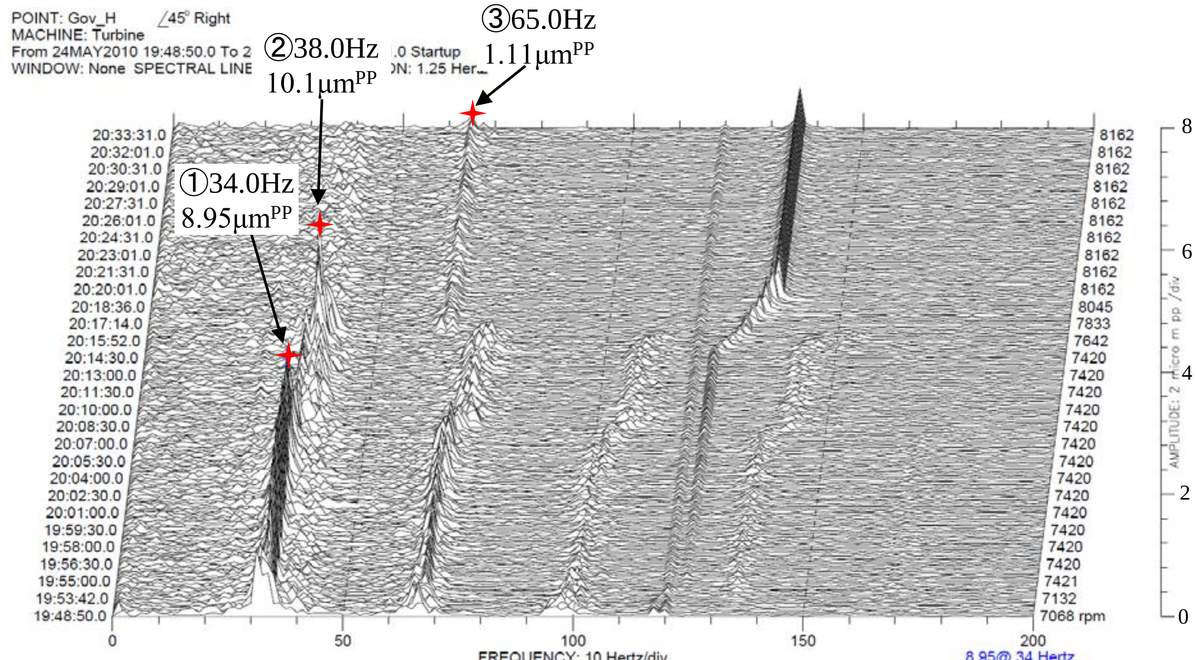
Case-1

Configuration of Bearings on Governor Side

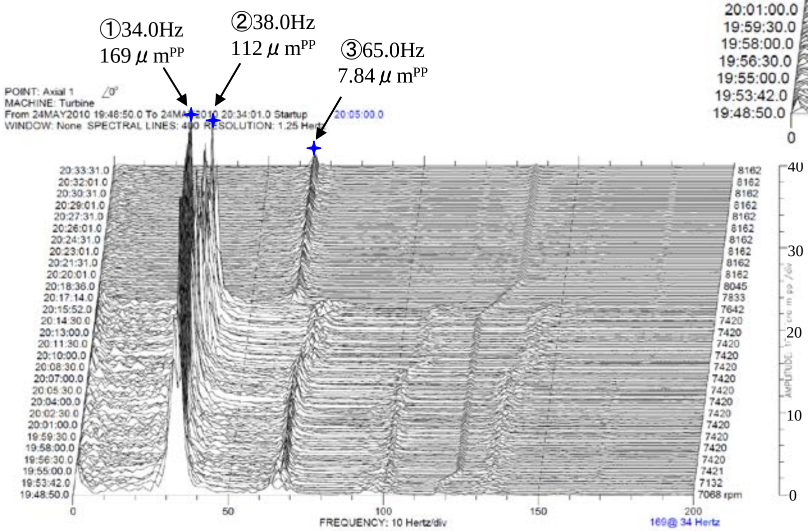


Coupled Sub-Synchronous Vibration Lateral and Axial Directions

Case-1



Water fall Plot for Lateral Vibration (Gov. Side)



Water fall Plot for Axial Vibration

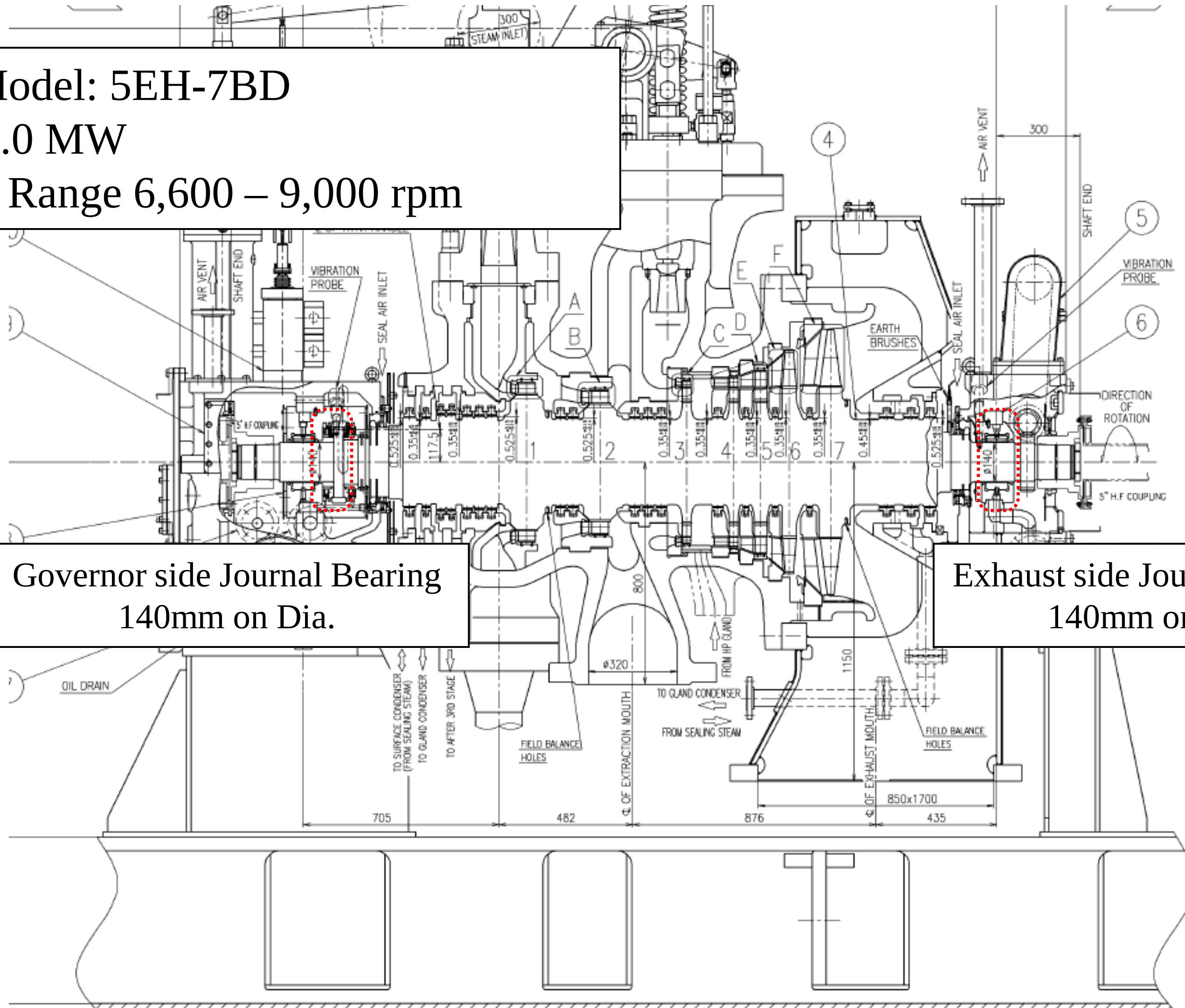
The same sub-synchronous frequency was observed for lateral and axial direction.

Coupled Sub-Synchronous Vibration Lateral and Axial Directions

Case-2

Another sub-synchronous vibration was observed during the shop mechanical running test of the following steam turbine.

Turbine Model: 5EH-7BD
Power: 43.0 MW
Operating Range 6,600 – 9,000 rpm

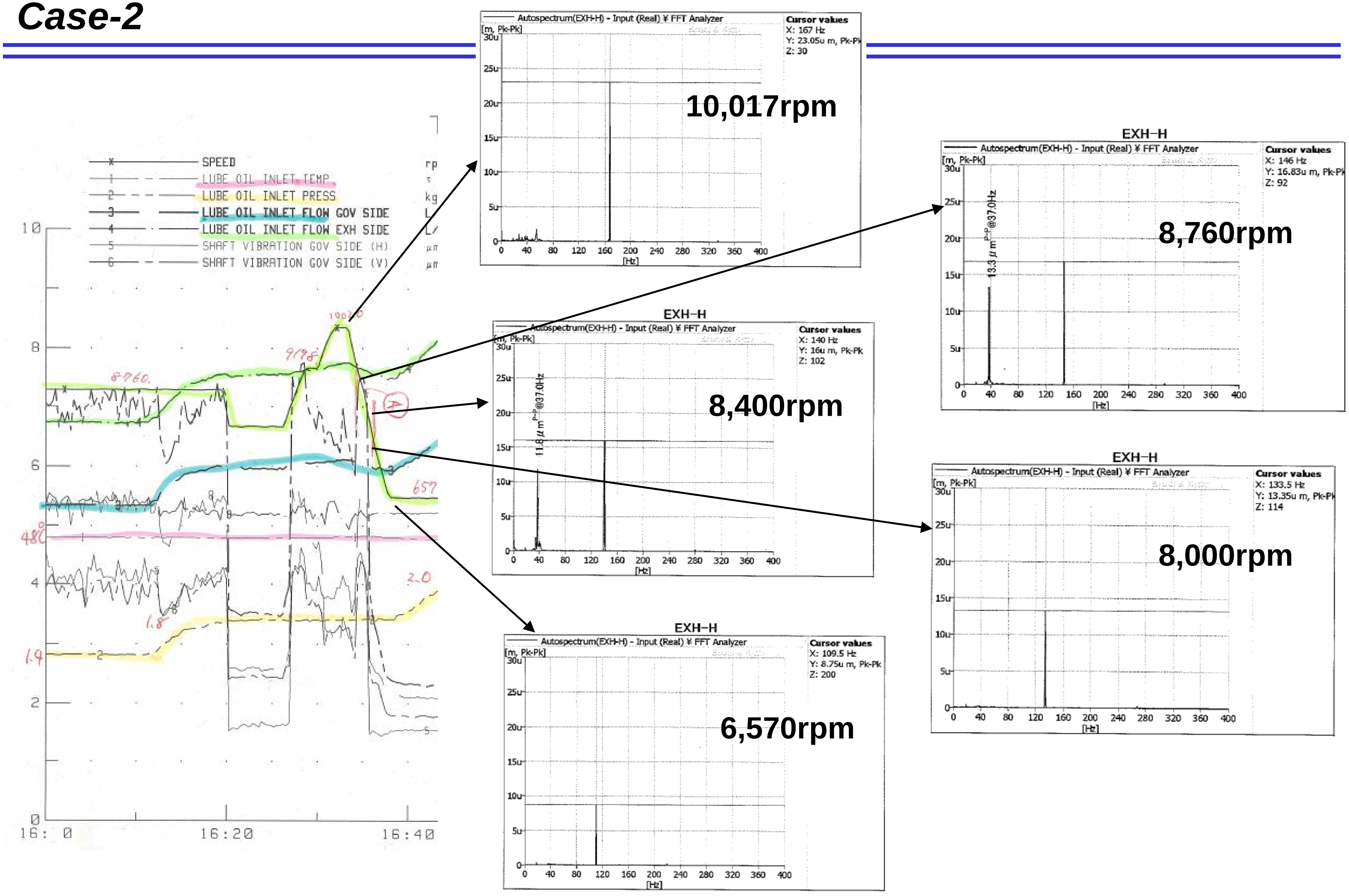


Governor side Journal Bearing
140mm on Dia.

Exhaust side Journal Bearing
140mm on Dia.

Coupled Sub-Synchronous Vibration Lateral and Axial Directions

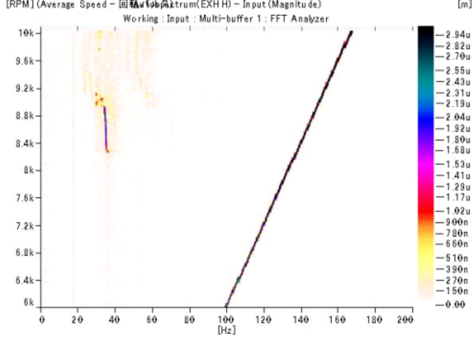
Case-2



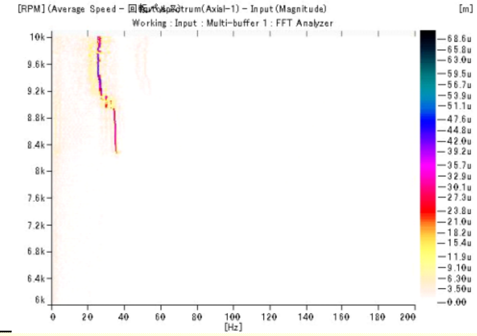
Coupled Sub-Synchronous Vibration Lateral and Axial Directions

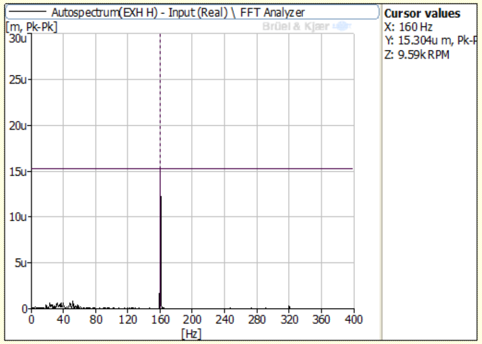
Case-2

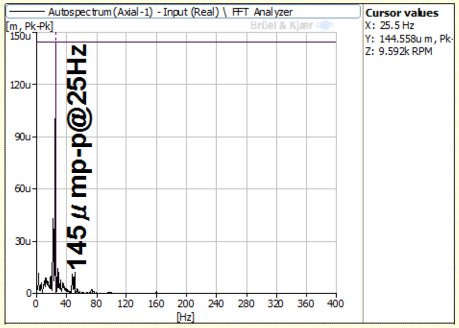
Lateral Direction

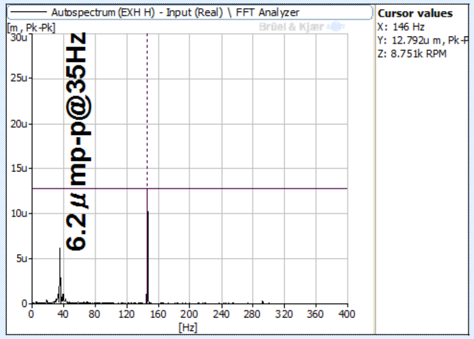


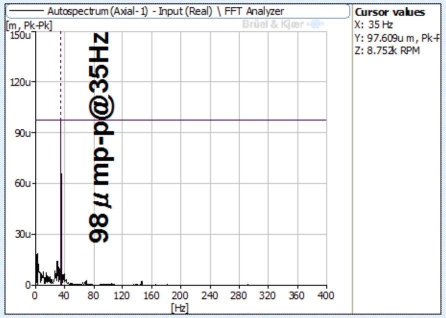
Axial Direction











The same sub-synchronous frequency was observed for lateral and axial direction.

Coupled Sub-Synchronous Vibration Lateral and Axial Directions

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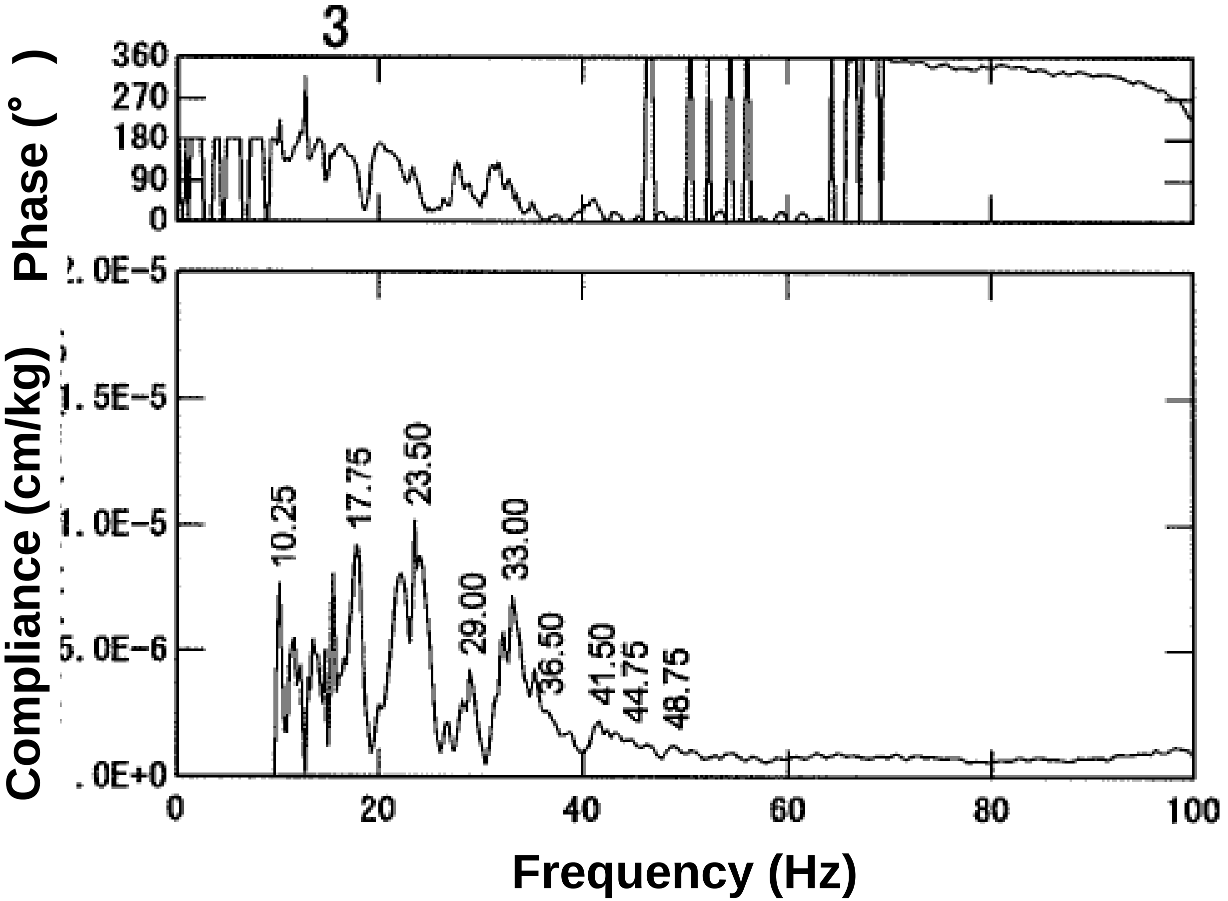
Case-2

Hammering Test for Pedestal



Natural Frequency of 35 - 40Hz could not be found.

Coupled Sub-Synchronous Vibration Lateral and Axial Directions

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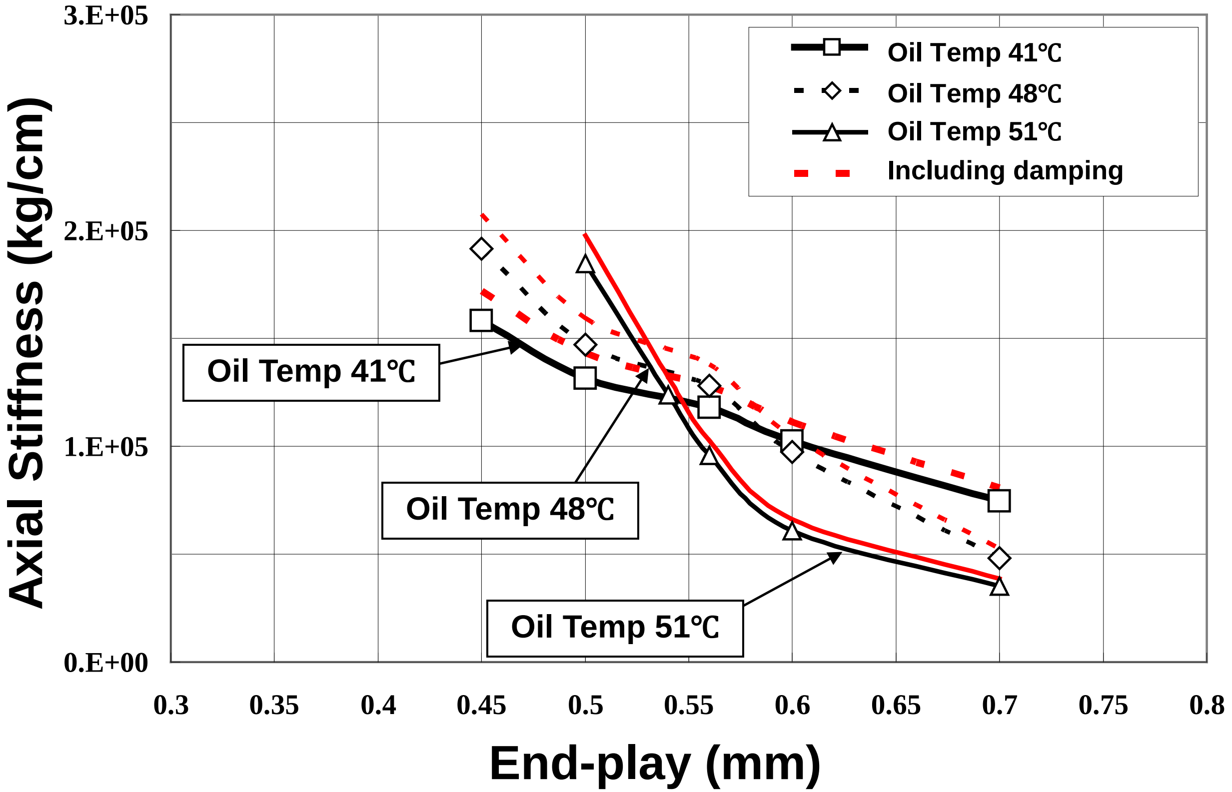
Case-2

Simulation Results of Thrust Bearing Stiffness



Axial Natural Frequency

$$m \approx 2{,}000 \ \ kg$$

$$k \approx 1 \times 10^{5} \ \ kgf\,/\,cm$$

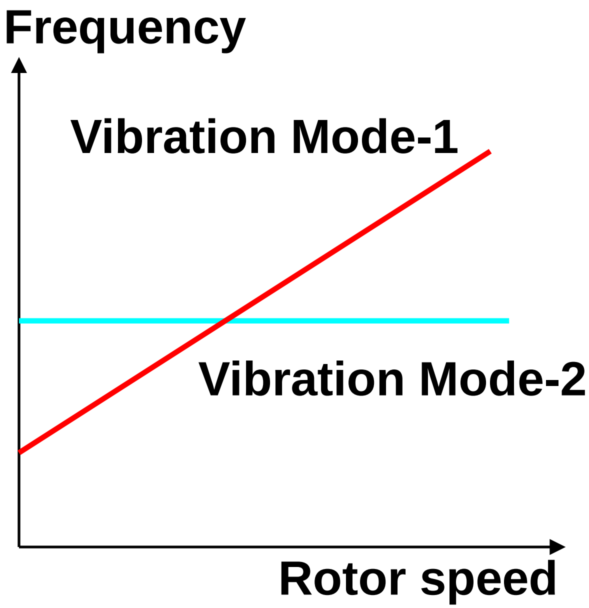
$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

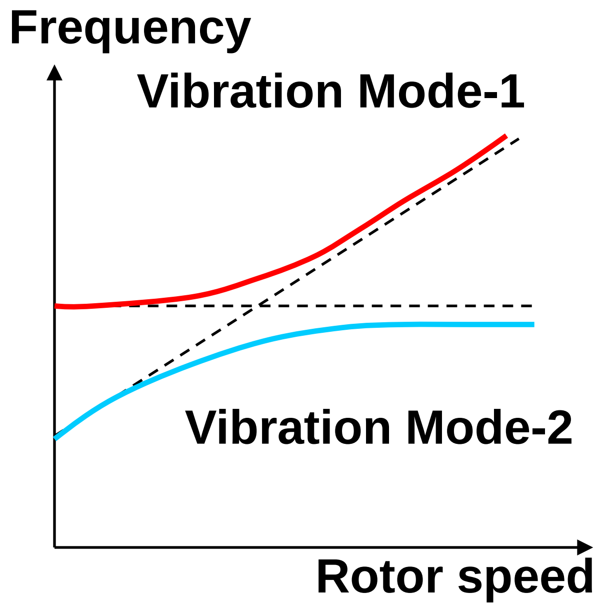
$$= \frac{1}{2\pi}\sqrt{\frac{1 \times 10^{5} \times 980}{2{,}000}}$$

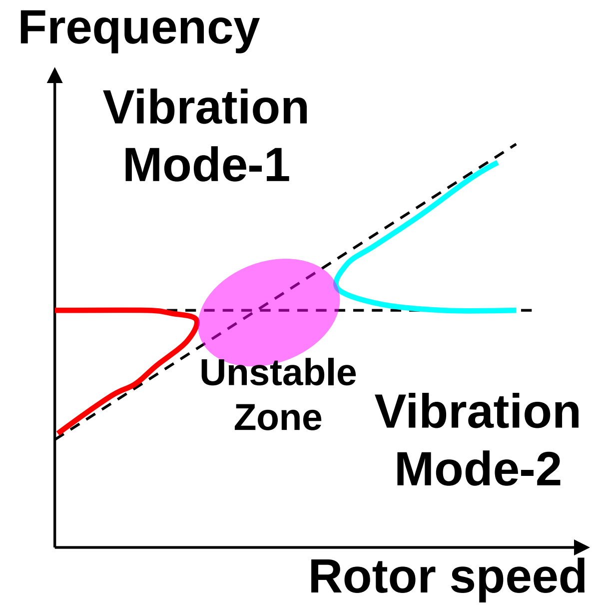
$$= 35.2 \ \ Hz$$

Agreed with sub-synchronous vibration frequency.

Unstable Coupled Vibration of Lateral and Axial Directions







No Energy Interaction

General

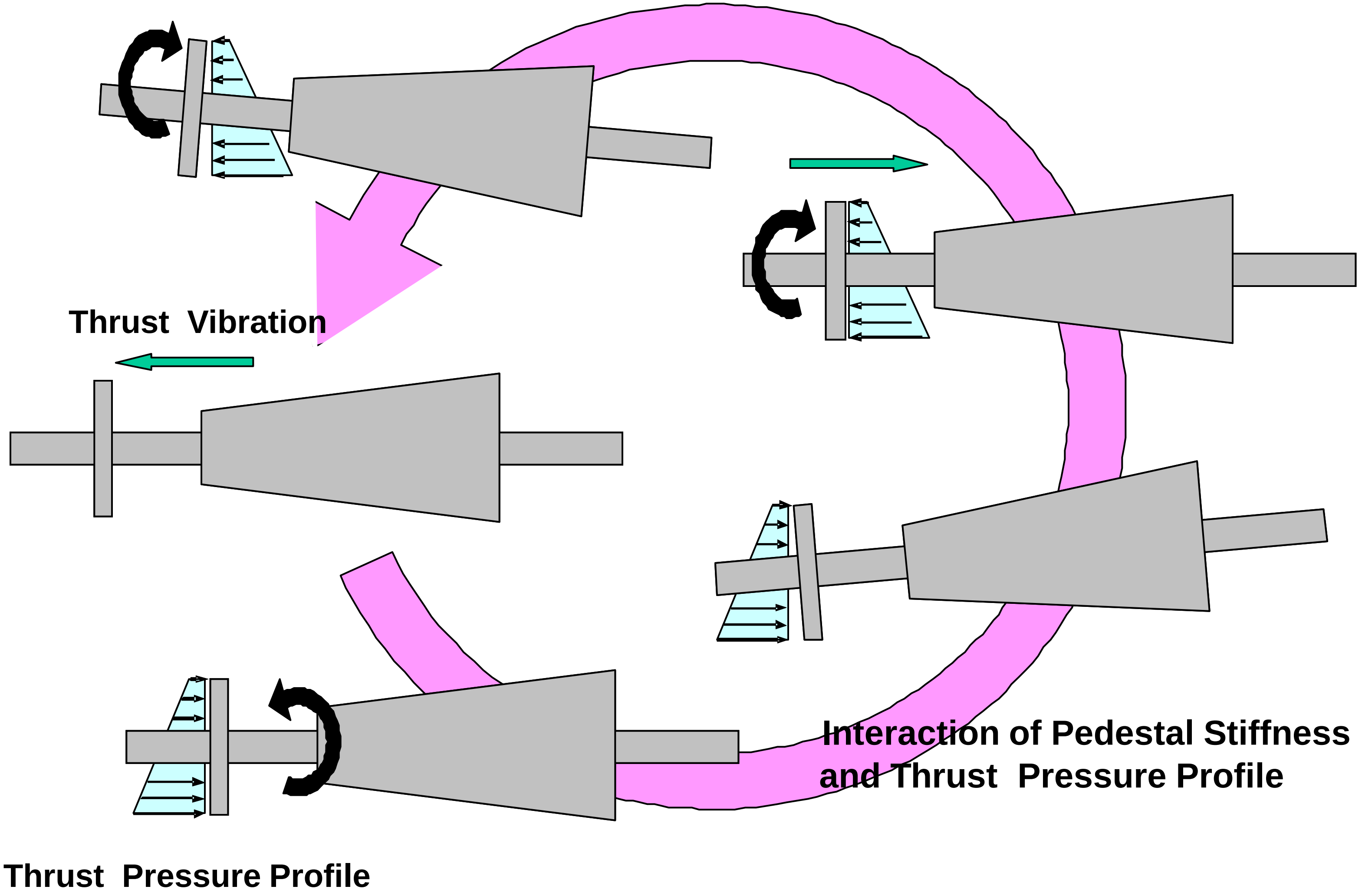
Stable

Coupled Vibration

Unstable

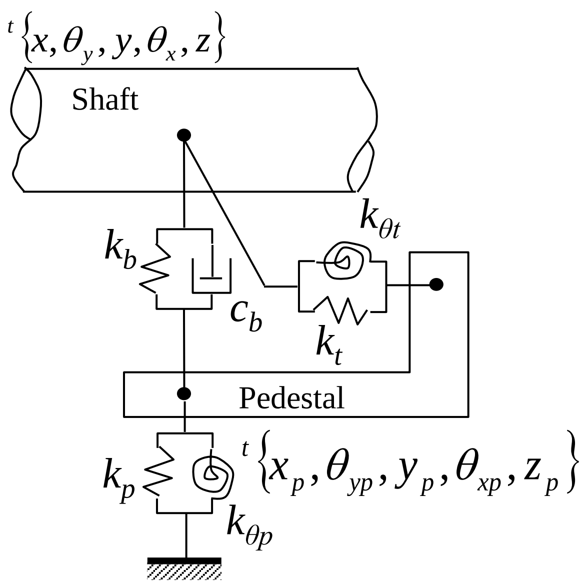
Coupled Vibration

Image for Coupled Vibration of Lateral and Axial Directions



Model for Coupled Vibration of Lateral and Axial Directions

Axial direction freedom is considered in the rotor and pedestal support system.



k_b : *radial bearing stiffness*

c_b : *radial bearing damping*

$k_{\theta t}$: *Thrust bearing angular stiffness*

k_{zt} : *Thrust bearing stiffness*

k_p : *pedestal stiffness*

$k_{\theta p}$: *pedestal angular stiffness*

-Matrix of shaft stiffness

$$\begin{Bmatrix} F \\ M \end{Bmatrix} = \begin{bmatrix} k_{11} & k_{12} \\ k_{21} & k_{22} \end{bmatrix} \begin{Bmatrix} x \\ \theta \end{Bmatrix} = K_r \begin{Bmatrix} x \\ \theta \end{Bmatrix}$$

-Matrix of radial bearing stiffness

$$\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = \begin{bmatrix} k_{xx} & \\ & k_{yy} \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} = K_b x$$

-Matrix of thrust bearing stiffness

$$\begin{Bmatrix} M_y \\ M_x \\ F_z \end{Bmatrix} = \begin{bmatrix} k_{\theta txx} & k_{\theta txy} & \\ k_{\theta tyx} & k_{\theta tyy} & \\ & & k_{tz} \end{bmatrix} \begin{Bmatrix} \theta_y \\ \theta_x \\ z \end{Bmatrix} = K_t x$$

-As a system characteristic matrix

$$\begin{Bmatrix} F_{xp} \\ M_{yp} \\ F_{yp} \\ M_{xp} \\ F_{zp} \end{Bmatrix} = \begin{bmatrix} k_{pxx} & & & & \\ & k_{\theta pyy} & & & k_{\theta pyz} \\ & & k_{pyy} & & \\ & & & k_{\theta pxx} & k_{\theta pxz} \\ & k_{\theta pzy} & & k_{\theta pzx} & k_{pz} \end{bmatrix} \begin{Bmatrix} x_p \\ \theta_{yp} \\ y_p \\ \theta_{xp} \\ z_p \end{Bmatrix}$$

Model for Coupled Vibration of Lateral and Axial Directions

The eigenvalue problem is solved.

$$
\begin{bmatrix}
m &&&&&&&&& \\
& m &&&&&&&& \\
&& m_p &&&&&&& \\
&&& m_p &&&&&& \\
&&&& I_d &&&&& \\
&&&&& I_d &&&& \\
&&&&&& I_{dp} &&& \\
&&&&&&& I_{dp} && \\
&&&&&&&& m & \\
&&&&&&&&& m_p
\end{bmatrix}
\begin{Bmatrix} \ddot{x} \\ \ddot{y} \\ \ddot{x}_p \\ \ddot{y}_p \\ \ddot{\theta}_y \\ \ddot{\theta}_x \\ \ddot{\theta}_{yp} \\ \ddot{\theta}_{xp} \\ \ddot{z} \\ \ddot{z}_p \end{Bmatrix}
+
\begin{bmatrix}
+k_{bxx} & & -k_{bxx} & & & & & & & \\
& +k_{byy} & & -k_{byy} & & & & & & \\
-k_{bxx} & & k_{bxx} & & & & & & & \\
& -k_{byy} & & k_{byy} & & & & & & \\
& & & & +k_{\theta tyy} & k_{\theta tyx} & -k_{\theta tyy} & -k_{\theta tyx} & & \\
& & & & k_{\theta txy} & +k_{\theta txx} & -k_{\theta txy} & -k_{\theta txx} & & \\
& & & & -k_{\theta tyy} & -k_{\theta tyx} & k_{\theta tyy} & k_{\theta tyx} & & k_{\theta yzp} \\
& & & & -k_{\theta txy} & -k_{\theta txx} & k_{\theta txy} & k_{\theta txx} & & k_{\theta xzp} \\
& & & & & & & & k_z & -k_z \\
& & & & & & k_{\theta zyp} & k_{\theta zxp} & -k_z & k_{zp}
\end{bmatrix}
\begin{Bmatrix} x \\ y \\ x_p \\ y_p \\ \theta_y \\ \theta_x \\ \theta_{yp} \\ \theta_{xp} \\ z \\ z_p \end{Bmatrix}
= 0
$$

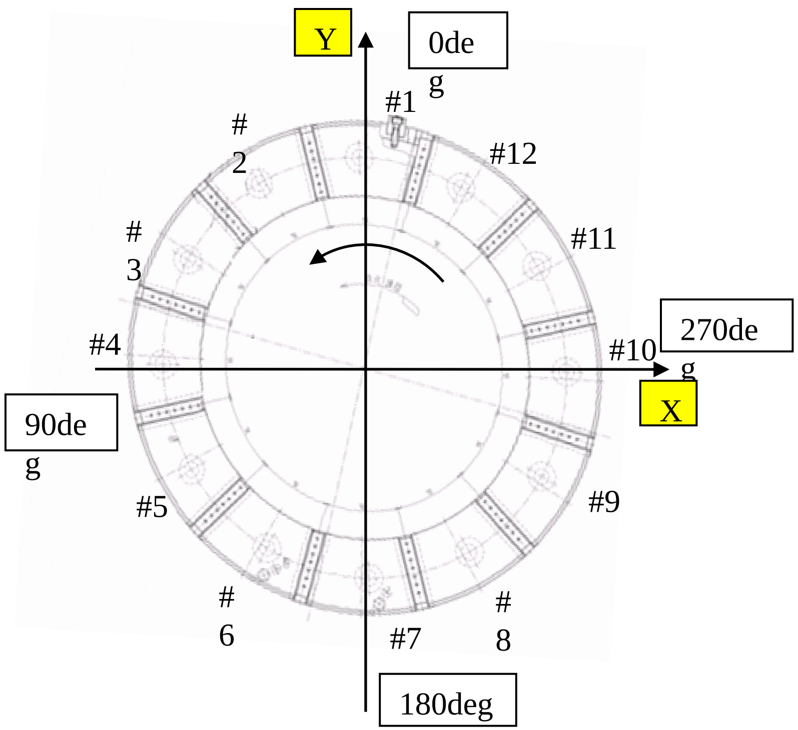
By simply expressed,

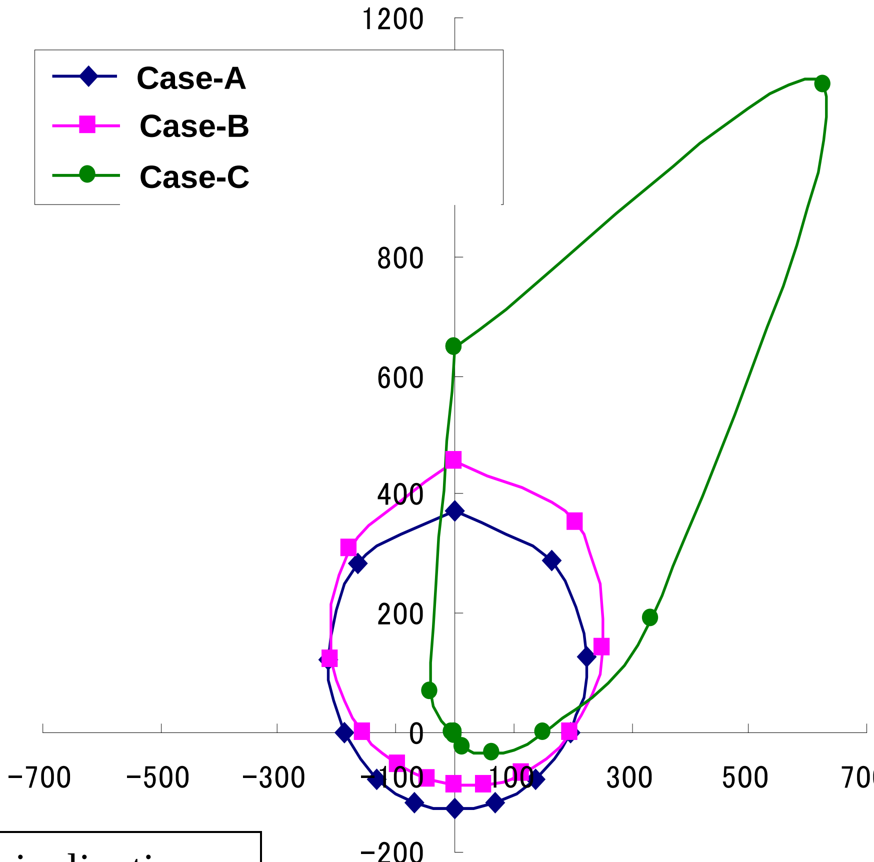
Symmetry

$$
\begin{bmatrix}
\boldsymbol{m} & & & \\
& \boldsymbol{I} & & \\
& & m & \\
& & & m_p
\end{bmatrix}
\begin{Bmatrix} \ddot{\boldsymbol{X}} \\ \ddot{\boldsymbol{\Theta}} \\ \ddot{z} \\ \ddot{z}_p \end{Bmatrix}
+
\begin{bmatrix}
\boldsymbol{K}_b & & & \\
& \boldsymbol{K}_{th} & & \boldsymbol{K}_{\theta M} \\
& & k_z & -k_z \\
& \boldsymbol{K}_{\theta M} & -k_z & k_{zp}
\end{bmatrix}
\begin{Bmatrix} \boldsymbol{X} \\ \boldsymbol{\Theta} \\ z \\ z_p \end{Bmatrix}
= \boldsymbol{0}
$$

Only K_{th} is related to the stability because other terms are symmetry.

Simulation Result for Thrust Load Distribution of Pads





Load Distribution of Pads

Case-A: Pads gnathonically move to the thrust disc inclination.
Case-B: Pads inclination is fixed at the stationary state.
Case-C: Pads are fixed as a flat face.

The cross coupling stiffness becomes larger if the pad static equilibrium are broken up. It would become the driving force of the radial/ axial coupled vibration.

Summary

During shop mechanical running test of steam turbine, radial sub-synchronous vibration was observed. And, at the same sub-synchronous frequency, the axial vibration was also observed.

It was confirmed that the sub-synchronous frequency agrees with the axial natural frequency of the rotor.

Rotor support system was modeled to investigate this interaction of axial and radial direction vibration by considering the axial direction freedom.

As a result of the simulation, the cross coupling stiffness becomes larger if the pad static equilibrium are broken up.

It was confirmed that it becomes the driving force of the radial/ axial coupled vibration.

It is expected that the coupled sub-synchronous vibration of lateral and axial directions can be predicted and evaluated by the sumilation.